# LESSONS LEARNED, CLIMATIC CHAMBERS, MATERIEL TEST FACILITY

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## INTRODUCTION

The Multipurpose Chamber (MPC) is a test chamber within the Materiel Test Facility at Dugway Proving Ground in Utah. The chamber was designed to be capable of chemical agent tests at various internal environmental conditions between 125 degrees F, 95% relative humidity, and minus 65 degrees F. The system design indicated that 50 tons of cooling capacity would be sufficient to cool the chamber to a temperature of minus 65 degrees F. The initial tests of the refrigeration system during the summer of 1995 indicated a minimum attainable temperature of only minus 35 degrees F.

The chamber is 50' x 50' and is 30' tall (Figure 1). The interior of the chamber walls and ceiling are constructed of welded 14 gage stainless steel panels. The chamber's 8" thick concrete floor slab is covered with welded 1/8" stainless steel plate. The chamber's walls, ceiling, floor, doors, piping, and conduit penetrations are all welded, gasketed, and/or calked in order to be airtight and suitable for chemical agent tests and decontamination wash down. The walls and ceiling are insulated with a 4" thick metal faced panel system which has an effective insulating value of R=32. The concrete floor is insulated underneath with 6" of insulation for an insulating value of R=30. The sliding door is insulated to R=30 but the effective R-value may be closer to R=15 due to the thermal bridging of the door's structural skeleton.

The chamber's construction, cooling load, and refrigeration system were analyzed to determine if the chamber cooling load was greater than anticipated and/or if the refrigeration system was providing less cooling than required. This study revealed a number of problems with the chamber construction and the design of the refrigeration system.

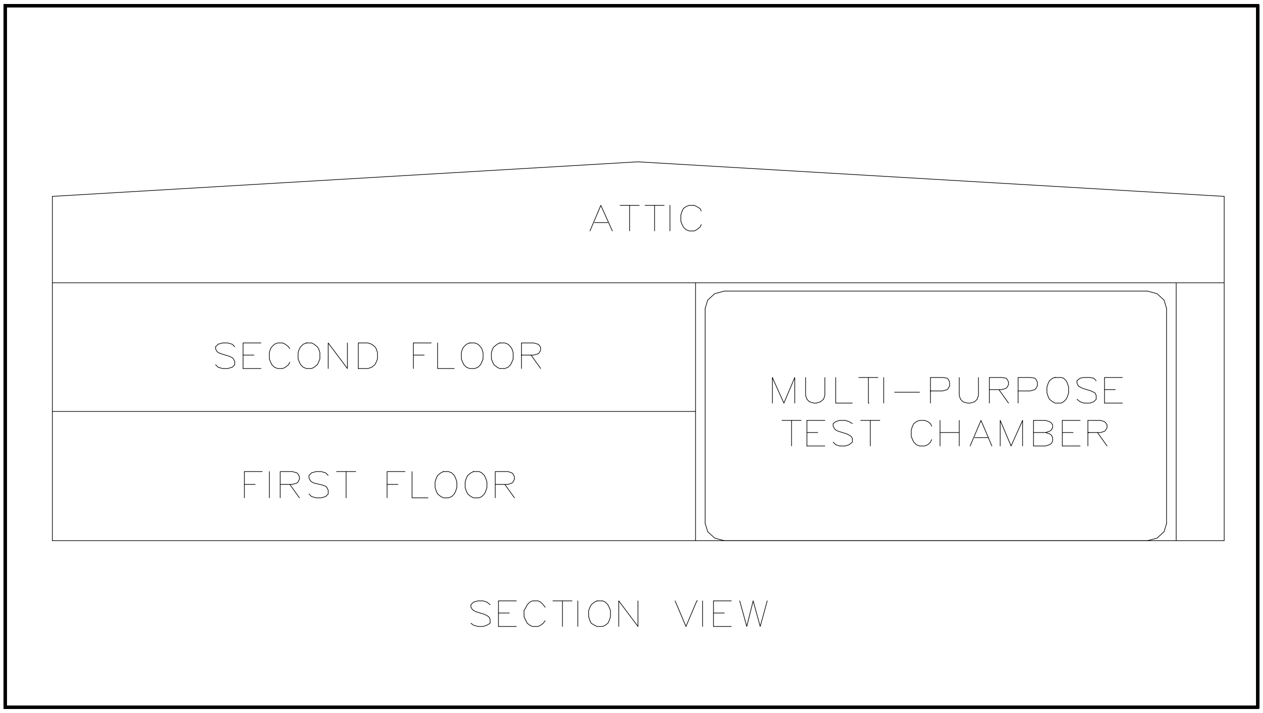


Figure 1

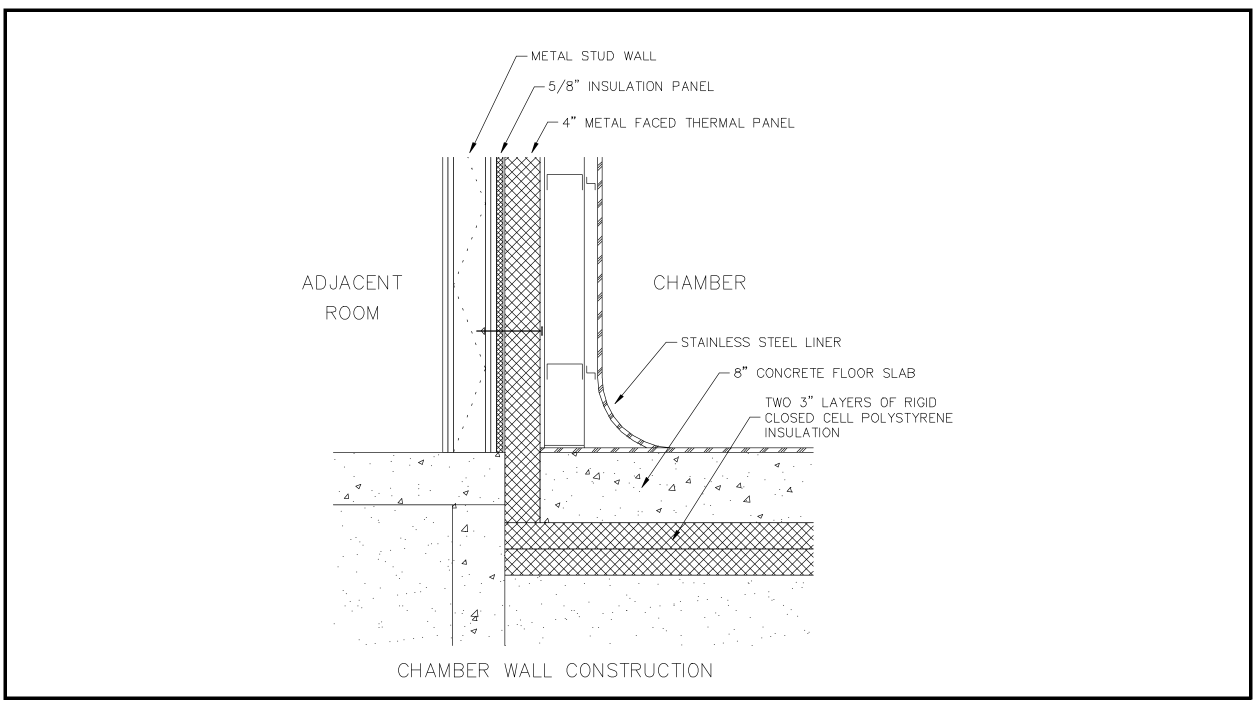


Figure 2

## REFRIGERATION SYSTEM CONFIGURATION

The refrigeration system consists of two water cooled screw chillers, an accumulator, two liquid refrigerant pumps, and two low temperature fan coils (Figure 3).

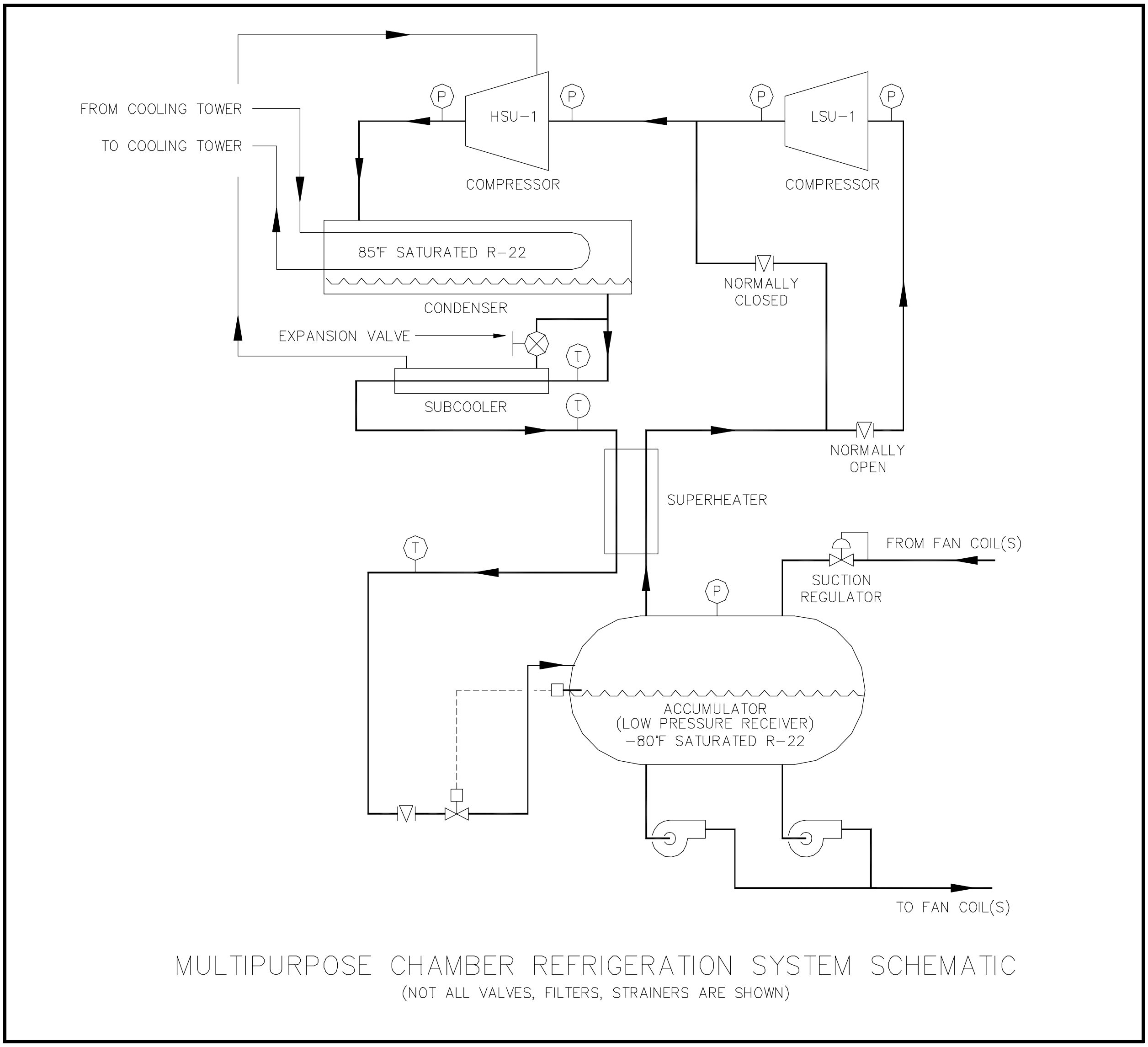


Figure 3

The refrigeration system was performance specified to produce 50 tons of cooling capacity with one pump supplying minus 80 degree F liquid R-22 to one of the fan coils. The fan coil schedule lists a refrigerant supply temperature of minus 80 degrees F, refrigerant return temperature of minus 70 degrees F, and a fan coil capacity of 50 tons (600,000 btu/hr). A required fan coil refrigerant flow of 319 gpm was calculated based on the 10 degree F temperature difference, 50 ton capacity, and the specific heat of R-22. During an August 1995 system test a flow of 280 $\pm$ 20 gpm was determined based on the differential pressure across the pump and the pump performance curve. Operating both refrigerant pumps would increase refrigerant flow only slightly because the pump curve is very flat in this region. The original contract required the pump(s) to have a shutoff head 20% greater than design head and 15 hp motors. Although the installed pumps have 20 hp motors the shutoff head is only 6% above the current operating head. The pump motors are hermetic and are cooled by the refrigerant. The pump motor heat is a cooling load on the refrigeration system but not on the chamber.

**FAN COIL CAPACITY**

The fan coils inside the chamber were specified to provide 50 tons of cooling capacity with liquid R-22 refrigerant supplied at minus 80 degrees F and returning at minus 70 degrees F. The fan coil schedule incorrectly indicated an entering air temperature of minus 47.5 degrees F and a supply air temperature of minus 65 degrees F. To produce 50 tons of cooling in the chamber an entering air temperature of minus 65 degrees F and supply air temperature of minus 85 degrees F should have been specified in the fan coil schedule. The capacity of the fan coils with an entering air temperature of minus 65 degrees F and entering refrigerant temperature of minus 80 degrees F is approximately 23 tons. If the second fan coil was also operated, an additional motor load of 7.3 tons would increase the cooling load from 23 to 30 tons. The liquid R-22 refrigerant would be split between the two fan coils and would produce less than 30 tons of refrigeration.

A fan coil capacity of 48 tons was measured on August 9, 1995 during a system test with the chamber air temperature at minus 35 degrees F. The return line liquid temperature was determined by a pipe surface RTD temperature sensor. Since the RTD is not located in the liquid refrigerant it may not have provided an accurate temperature reading. The controls subcontractor estimated a 3 degree F differential between the return liquid temperature and the pipe surface temperature. So the actual capacity being delivered may have been only 39 tons.

## FAN COIL DEFROST CYCLE

The fan coil defrost cycle uses hot compressor gas to melt ice/frost from the 20 row, 4 fins per inch cooling coil after an air pressure differential is measured across the coil. At low chamber air temperature (-35 degrees F) defrost occurs about every 12 hours. When a fan coil shuts down to defrost the other fan coil starts and operates until it requires defrosting. One of the fan coils takes 30 minutes to defrost while the other only takes around 5 or 10 minutes. The control subcontractor theorized the liquid refrigerant returning from one fan coil may be backfilling the cooling coil on the other fan coil resulting in a longer defrost cycle. Since there is only one defrost timer it has been set at 30 minutes. During a 30 minute defrost cycle the chamber air temperature rises about 8 degrees F. The hot gas returning to the accumulator in the liquid return line may be restricting the liquid refrigerant flow returning from the other fan coil. During the approximate 12 hour fan coil operating time the accumulation of ice/frost on the 20 row cooling coil may also reduce the fan coil airflow below 29,000 cfm and result in reduced fan coil capacity.

## BACKPRESSURE CONTROL VALVE (SUCTION REGULATOR)

The refrigerant inside the Chamber is intended to be above atmospheric pressure at all times to prevent potential toxic test gases/fluids from leaking from the chamber into the refrigerant. The system was designed with a backpressure control valve (suction regulator) on the liquid return line from the fan coils to the accumulator to maintain a positive refrigerant pressure in the chamber refrigerant piping. The requirement for the backpressure control valve had been debated during construction and was initially not installed. Without a backpressure control valve the refrigerant pressure in the fan coil would fall below atmospheric pressure and a portion of the liquid refrigerant would vaporize. The 5" liquid return line from the fan coils was not sized for a mixture of liquid and vapor so refrigerant gas/liquid velocity and pressure drops would be higher than anticipated. A backpressure control valve (suction regulator) was subsequently installed on the liquid return line near the accumulator to maintain a positive pressure and liquid phase refrigerant in the liquid return lines.

**ACCUMULATOR**

The accumulator installed by the contractor was of insufficient pressure rating. The original contract specification required the accumulator to be suitable for the "maximum and minimum pressure or temperature encountered". The operating/non-operating pressures for the accumulator are between -10 psig and +225 psig. The installed accumulator was ASME rated for only 150 psig and had a pressure safety relief valve to relieve refrigerant if the pressure exceeded 150 psig (77 degree F saturated refrigerant R-22). The accumulator was subsequently retested and ASME certified for a pressure of 250 psig.

The compressors were losing crankcase oil to the refrigerant gas as expected but the oil recovery system on the accumulator was not returning the oil sufficiently. The approximately 350 gpm of refrigerant returning to the accumulator was determined to be causing violent mixing and foaming. The oil never had a chance to separate out and rise to the top of the refrigerant due to the agitation. Based on recommendations by a refrigeration specialist a coalescent oil filter was added to the discharge line of the second stage compressor. The coalescent oil filter resulted in little if any oil getting out into the system.

There was also a debate whether the accumulator is of sufficient size to contain all the refrigerant when the system is shut down. A 250 gallon high pressure receiver was subsequently added to the system between the condenser and the subcooler to hold additional liquid refrigerant. The accumulator serves as a vapor/liquid separator and as an expansion tank. The liquid refrigerant entering the accumulator from the fan coils and subcooler partially flashes to vapor due to the lower pressure in the accumulator. Refrigerant leaving the accumulator is either liquid pumped from the bottom, or vapor pulled from the top by one of the screw compressors. The vapor volume of the accumulator must be sufficient to reduce refrigerant velocity in the accumulator and minimize entrainment of liquid drops at the suction outlet. The density of R-22 varies from 70 lbs/ft$^3$ at 105 degrees F to 92 lbs/ft$^3$ at minus 85 degrees F. The accumulator liquid level should fluctuate mainly with refrigerant expansion/contraction due to temperature changes.

## CHAMBER COOLING LOAD CALCULATIONS

The total cooling capacity required to maintain the chamber at minus 65 degrees F was calculated to be approximately 23 tons (Figure 4). The envelope (floor, walls, ceiling, etc.) transmission cooling loads are small compared to the fan coil motor heat gain (7.3 tons) and infiltration load (10.4 tons). The most likely cause for excessive heat gain would be from unexpected infiltration above the 625 cfm assumed by the designer. The pollution abatement system operates to exhaust air from the chamber and create a negative 0.5 inches of water column static pressure. The chamber’s supply air handler is not intended to operate during the minus 65 degrees F test. The only infiltration air was expected to enter through the closed outside air louver of the supply air handler (measured at 310 cfm).

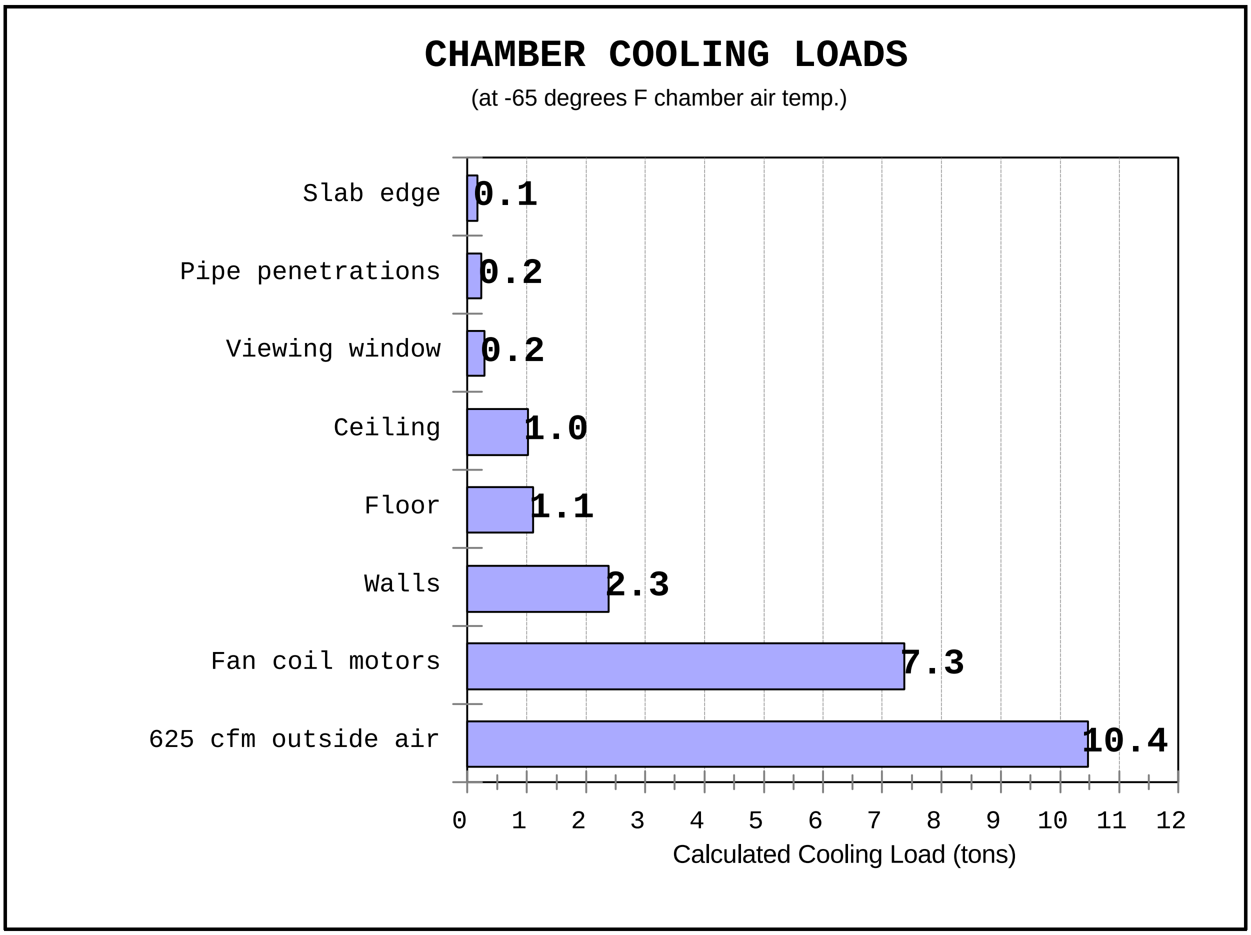


Figure 4

## INTERSTITIAL SPACE/VAPOR BARRIER

Cold temperature testing of the chamber resulted in substantial water drainage in the interstitial space between the stainless steel liner and metal faced thermal panels. The water was the result of melting ice which had formed in the interstitial space when the chamber temperature is below 32 degrees F. The contract drawings intended for the interstitial space to be a dead air space but did not include requirements to be air tight or for a vapor barrier. The rooms surrounding the chamber are held at cascading static pressures to control air migration within the facility. The differing surrounding static pressures and the lack of a sealed interstitial space resulted in a continuous air flow through the interstitial space. The moisture carried by the air flow is deposited as condensation or ice on the stainless steel liner during cold chamber tests. The interstitial space condensation/ice is an unwanted cooling load and can also result in water/ice damage to the chamber wall assembly. Subsequent application of calking and spray foam has reduced the air flow through the interstitial space and amount of condensation/ice.

## CHAMBER COOL DOWN PROCESS

The capacity of the fan coils varies with the temperature difference between the supplied refrigerant and the chamber air temperature. The chamber’s conduction and infiltration cooling loads increase as the chamber air temperature drops. The heat stored in the chamber’s internal mass (concrete floor slab, steel liner, sixty ton army tank) is released as the air temperature drops. The chamber air temperature will decrease rapidly at first and then level off as conduction and infiltration loads increase and the internal mass heat is released. As the internal mass temperature equalizes with the chamber air temperature the internal mass heat release will decrease and become zero.

The cooling capacity of the refrigeration equipment decreases as the temperature difference between the condenser water and the accumulator refrigerant increases (the condenser water system provides cooling to other mechanical equipment so it is maintained at 75-80 degrees F).

Based on the decreasing chamber air temperature, decreasing cooling capacity of the fan coils and refrigeration system, increasing cooling loads, and internal mass heat release the chamber cool down process is estimated to be as shown in Figure 5.

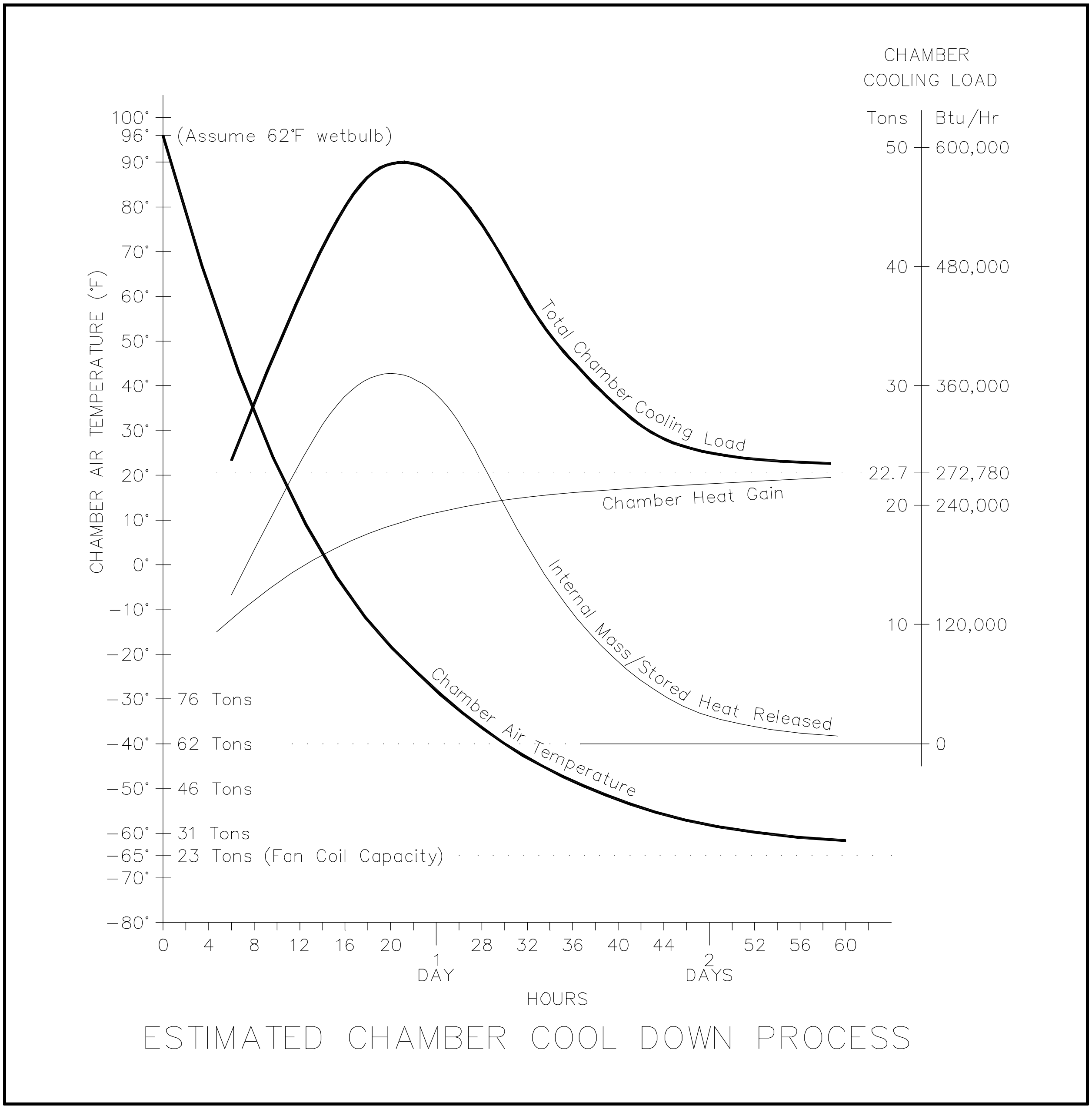


Figure 5

The vertical graph on the left side of Figure 5 includes fan coil cooling capacities between minus 30 F (76 tons) and minus 65 F (23 tons). The fan coil cooling capacities are based on a capacity graph from the A-E which charts fan coil capacity vs. chamber air temperature (with refrigerant supplied at minus 80 F). As the chamber air temperature approaches the temperature of the refrigerant the capacity of the fan coil decreases.

The vertical graph on the right side of Figure 5 indicates the chamber cooling load (Internal Mass/Stored Heat Release + Chamber Heat Gain = Total Chamber Cooling Load) in tons and Btu/hr.

Note that Figure 5 indicates a theoretical cool down process (not a steady state condition) based on calculated cooling loads, sufficient gpm of minus 80 degree F refrigerant, and the A-E’s fan coil capacity graph.

**CONCLUSION**

The facilities coldest test requirements were revised to minus 35 degrees F rather than minus 65 degrees F. The modifications to the refrigeration system and the reduced interstitial space air infiltration have resulted in a sustainable cold chamber temperature of minus 42 degrees F which meets the current requirements for the facility.

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